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(19)

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(54) SCREW COMPRESSOR

(71) We, VEB KUEHLAUTOMAT, a corporation organised under the laws of the German Democratic Republic, of 15—27 Segelfliegerdamm, Berlin, Germany, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to screw compressors for compressing gaseous media including; a pair of parallel rotors mounted for rotation each in a respective one of a pair of mutually intersecting parallel bores within a housing and having intermeshing helical teeth; a pair of flat faced covers closing the axial ends of the housing, one of which covers includes an inlet, and the other of which covers includes an outlet for the gaseous medium.

In known screw compressors, these covers also contain the support bearings for the rotor spindles and thrust bearings for axially fixing the rotors and adapted to take up axial thrust of the rotors directed from the delivery sides to the intake. For reasons of space, the supporting bearings for rotor spindles are usually journal bearings because the inlet and outlet orifices, generally segments of a circle centred on the bearings should extend, if possible, to the bottom of the rotor toothing. For reasons of high efficiency the meshing rotors are fitted with very little clearance, with regard to their outer diameter and length in relation to the inner profile of the housing and the flat faces of the covers, and are provided with an ample, continuously monitored supply of oil. In order to prevent narrow sealing strips frequently used on the toothed peripheral faces of the rotors from fouling the bores within the housing and causing progressive wear of the support bearings, the bores receiving the rotors and bores for the associated bearings must be very accurately aligned to achieve the clearance necessary to produce a high degree of efficiency. In order to achieve the clearance necessary for high efficiency between the delivery side axial faces of the rotors and the flat face of the cover including the outlet, thrust bearings for the rotors must be accurately adjusted and this requires additional equipment. Furthermore, the preset clearance increases during the operation of the compressor, due to deformation of the parts of the bearing under the action of the operating forces and wear of the bearing itself.

Furthermore, the support bearings necessitate an oil system supplying them with an automatically monitored amount of oil and using comparatively expensive equipment.

In the case of compressors operated with safety refrigerants, the oil coming from the intake side bearings is supplied to the intake side of the compressor with the evaporating refrigerant causes deterioration of the latter. In addition, the bearings form a substantial proportion of the length of the rotors and increase thereby the space requirements and the weight of the whole compressor.

The present invention may overcome these disadvantages by eliminating the need for the known support bearings whilst keeping the clearance between the delivery side and face of the rotor and the face of the cover having the outlet substantially small and constant throughout the whole working life of the compressor irrespective of the operating conditions.

The present invention provides a screw compressor for compressing gaseous media including; a pair of parallel rotors mounted for rotation each in a respective one of a pair of mutually intersecting parallel bores within a housing and having intermeshing helical teeth; a pair of flat faced covers closing the axial ends of the housing, one of which covers includes an inlet, and the other of which covers includes an outlet for the gaseous medium; peripheral gaps between the radially outer faces of the teeth of both rotors and the associated bore, said peripheral gaps each being at least partially wedge-shaped in a plane perpendicular to the rotor axes and each having a maximum

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radial width at a side of the gap which is leading relative to the rotary movement of the associated tooth for producing hydrodynamic radial support for the rotor; axial wedge-shaped gaps between radially extending inclined end surfaces of each rotor and said other cover, said axial gaps each being of maximum axial width at a side of the gap which is leading relative to the rotary movement of the associated rotor for producing hydrodynamic axial support of the rotor; means for providing axial thrust upon the rotors towards said other cover, said thrust being greater than the operational rotor thrust in the reverse direction; and an axial centre bore in at least one of the rotors, said centre bore having internal splines through which the compressor can be driven.

The invention eliminates the need for support bearings mounted on the covers. Stub shafts may, however be provided on the rotors, but such shafts may be dispensed with to provide a more easily manufactured compressor and to enable the rotors to be entirely self adjusting.

The means for providing the axial thrust upon the rotors will counteract the operational thrust in the reverse direction caused by the gas pressure on the delivery side of the rotors, and will prevent the rotors from lifting off the said other cover under the action of the gas pressure and from being pressed onto the said one cover. Such thrust may suitably act through stub shafts of the rotors. It will be understood, however, that the hydrodynamic effect of the axial gaps will move the rotors out of actual contact with the said other cover.

The means for producing axial thrust upon the rotors may comprise a pair of thrust bearings coupled each to a respective rotor and movable each in a respective pressure chamber by a pressure medium, said pressure medium being lubricating oil and/or a part of the gaseous medium delivered to the compressor. Where lubricating oil is used, its pressure is controlled so that the thrust is always higher than the axial thrust produced by the gas pressure and acting on the rotors in the direction towards the suction side. The lubricant pressure or the gas pressure is suitably applied through known pressure chambers, acting on the revolving rotors each through a conventional thrust bearing.

Due to these devices, the stub shafts leading to hitherto used thrust bearings may be omitted. The thrust bearings mounted between the pressure chambers and the rotor shafts may have a worse quality than present angular bearings because the axial adjustment of the rotors takes place automatically by the hydrodynamic effect of the slightly inclined delivery side axial faces. Wear of the thrust bearing remains ineffective with regard to the end play of the rotors relative to the flat flanges of the cover.

The invention further proposes to provide an axial centre bore in one of the rotors with inner splines to provide the drive; the inner splines meshing with the corresponding outer splines on a shaft passing through a self-adjusting stuffing box in one of the two covers, and connected at its outer end, to a drive flange mounted rotatably on one of the covers. Even a small clearance in the interengaging splines ensures self-adjustment of the driven rotor within the surrounding inner surface of the housing.

The mounting of the rotors may substantially reduce the material requirements for the rotors, eliminate support bearings, and result in a saving in production costs. Also, the costs of the assembly of the compressor are smaller because the adjustment of the axial face clearance of the rotor in the housing can be eliminated. The need for precision finishing of the housing is reduced by the self-adjusting character of the rotor, and the fact that all surfaces affecting the position of the rotors relative to each other and to the housing are within the housing. Owing to the wear extending uniformly over the whole length of the rotor, bending of the rotors under the action of the gas forces may be substantially reduced and, rotors with a smaller rigidity than hitherto can be used to produce a larger delivery volume with the same housing dimensions as hitherto.

The invention will be further described, by way of example, with reference to a preferred screw compressor shown in the accompanying drawings, in which:—

Fig. 1 is an end view of the delivery side of the screw compressor with an axial cover detached:

Fig. 2 is a cross-section along the line A—A in Fig. 1;

Fig. 3 is a cross-section along the line B—B in Fig. 1;

Fig. 4 is a cross-section of the screw compressor along the line C—C in Fig. 1.

The end view of the delivery side of the screw compressor shown in Fig. 1 indicates a housing shell 1 with male and female rotors 3, 4 located in mutually intersecting parallel bores defined by an inner peripheral surface 2 of the housing and having helical, intermeshing toothing. The radially outer faces 5, 6 of the teeth of the rotors 3, 4 are so formed that they form narrow wedge-shaped gaps with the inner surface 2, the maximum radial width of each of which is at a side of the gap which is leading relative to the rotary movement 7, 8 of the associated one of the rotors 3, 4. The axial end faces 9, 10 of the rotors 3, 4 have slightly inclined radially extending surfaces 11, 12 forming, with a flat

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inner surface 19 of an end cover 20 of the housing wedge-shaped gaps 15, 16, the maximum axial width of each of which is at a side of the gap which is leading relative to the direction of the rotary movement 7, 8 of the associated one of the rotors 3, 4 (Figs. 2 and 3). A channel 14 serves to supply lubricating oil. Fig. 4 is a cross-section of the screw compressor along the line C—C in Fig. 1. The housing shell 1 with the inner surface 2 is closed on the intake side of the screw compressor by a flat inner surface 17 of the cover 18 containing an inlet port, and on the delivery side by the flat surface 19 of the cover 20 containing an outlet port. The rotor 3 has an axial centre bore 21 with internal splines 22. The cover 18 on the intake side is adapted to receive a pressure chamber or cylinder 23, A thrust bearing 5 is slidable in the cylinder 23 and is connected through a piston rod 24 to an end face of the rotor 15. The rotor 4 adjacent to the rotor 3 is also connected through a piston rod 24' to a thrust bearing 25' slidable in a pressure chamber or cylinder 23'. Outer splines 26 of a shaft 27 engage with the inner splines 22; the shaft 27 passes through a stuffing box 28 in the cover 20 and is connected at its outer end, having external splines 29, with a drive flange 30. The drive flange 30 is mounted in a bearing 31 located centrally in the cover 20.

The operation of the compressor is as follows:—

Prior to starting the rotors 3, 4, an oil pump, not shown, and forming part of the screw compressor plant, produces a flow of oil passing on the one hand through channel 14 to the chamber formed by the housing shell 1 and end covers, and the meshing toothing of the rotors 3, 4 and on the other hand through channels 32, 32' to the cylinders 23, 23', which produce an axial thrust directed towards the delivery side and acting on both rotors 3, 4 through piston rods 24, 24' and thrust bearings 25, 25'.

During the operation of the screw compressor, the rotor 3 is driven through the drive flange 30 and the shaft 27 and entrains the rotor 4 by flank contact of the toothing of the rotors 3, 4 shown in Fig. 1.

In view of the oil wetting of all rotor surfaces and internal housing surfaces, all wedge-shaped gaps produce immediately hydrodynamic support. This causes the tooth peripheral faces 5, 6 of both rotors 3, 4 to be lifted off the inner surface 2 of the housing 1, and the delivery side axial faces 9, 10 of the rotors to be separated from the flat surface 19 of the cover 20. End face contact between the suction side axial face and the surface 17 of the cover 18 on the suction side does not occur, because the axial force of the pressure cylinders 23, 23' is larger than the counteracting axial thrust of rotors, 3, 4 composed of the hydrodynamic supporting force and the gas pressure prevailing on the delivery side.

WHAT WE CLAIM IS:—

1. A screw compressor for compressing gaseous media including; a pair of parallel rotors mounted for rotation each in a respective one of a pair of mutually intersecting parallel bores within a housing and having intermeshing helical teeth; a pair of flat faced covers closing the axial ends of the housing, one of which covers includes an inlet, and the other of which covers includes an outlet for the gaseous medium; peripheral gaps between the radially outer faces of the teeth of both rotors and the associated bore, said peripheral gaps each being at least partially wedge-shaped in a plane perpendicular to the rotor axes and each having a maximum radial width at a side of the gap which is leading relative to the rotary movement of the associated tooth, for producing hydrodynamic radial support for the rotor; axial wedge-shaped gaps between radially extending inclined end surfaces of each rotor and said other cover, said axial gaps each being of maximum axial width at a side of the gap which is leading relative to the rotary movement of the associated rotor for producing hydrodynamic axial support for the rotor; means for providing axial thrust upon the rotors towards said other cover, said thrust being greater than the operational rotor thrust in the reverse direction; and an axial centre bore in at least one of the rotors, said centre bore having internal splines through which the compressor can be driven.

2. A screw compressor as claimed in Claim 1, wherein said means for providing axial thrust upon the rotors comprises a pair of thrust bearings coupled each to a respective rotor and movable each in a respective pressure chamber by a pressure medium, said pressure medium being lubricating oil and/or a part of the gaseous medium delivered to the compressor.

3. A screw compressor as claimed in Claim 1 or claim 2, wherein an axial centre bore is provided in one of the rotors, a drive shaft is received within said centre bore, external splines on the drive shaft engage the internal splines of the centre bore, and the drive shaft passes through a stuffing box mounted in one of the covers and is rotatively coupled to a flange mounted in a bearing on the cover.

4. A screw compressor substantially as herein described with reference to the accompanying drawings.

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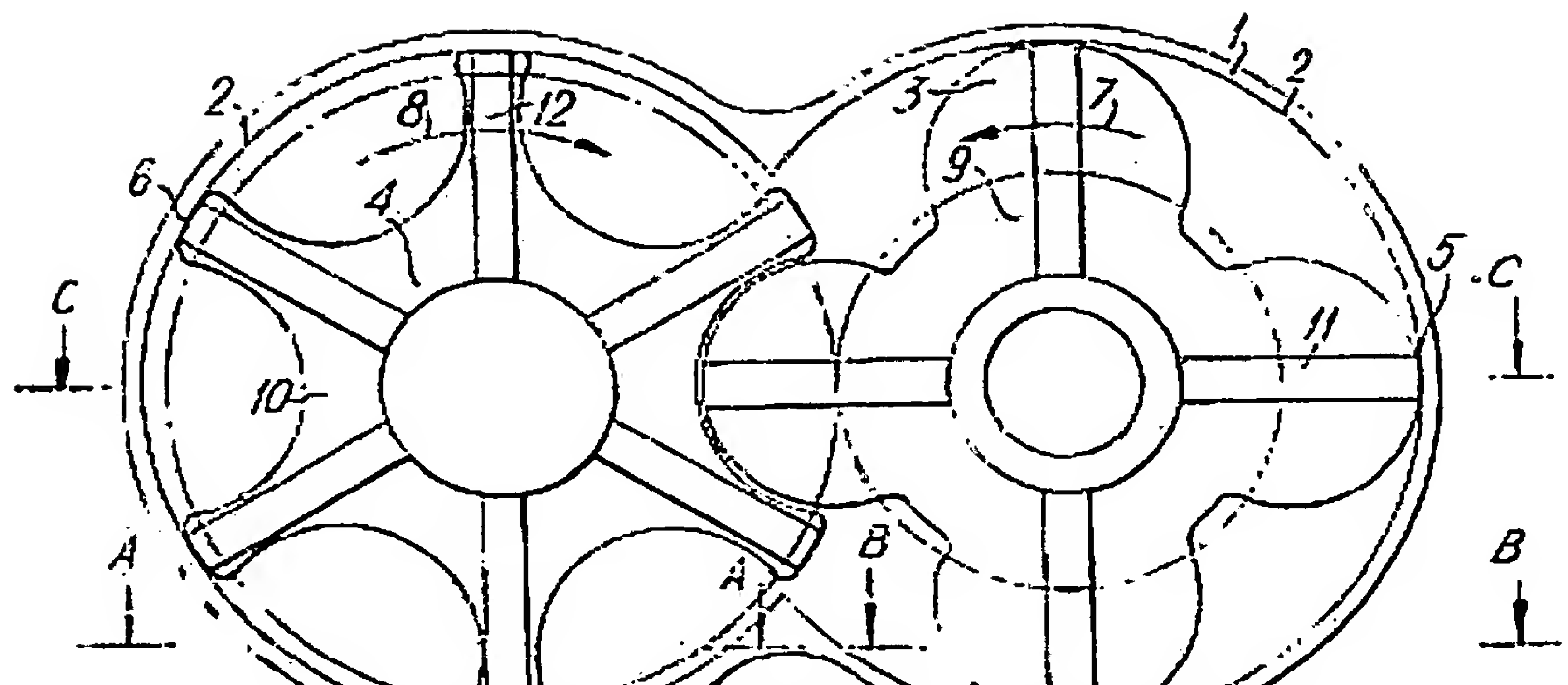

Fig.1.

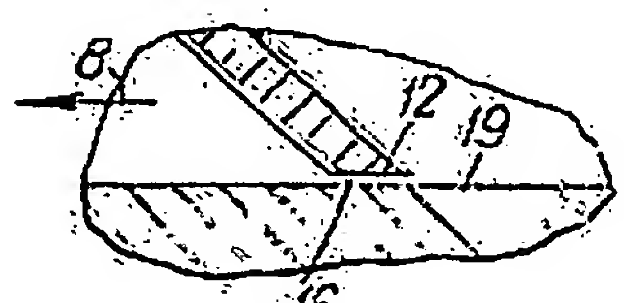

Fig. 2.

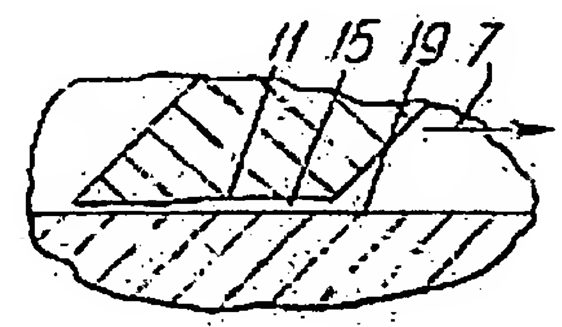

Fig.3.

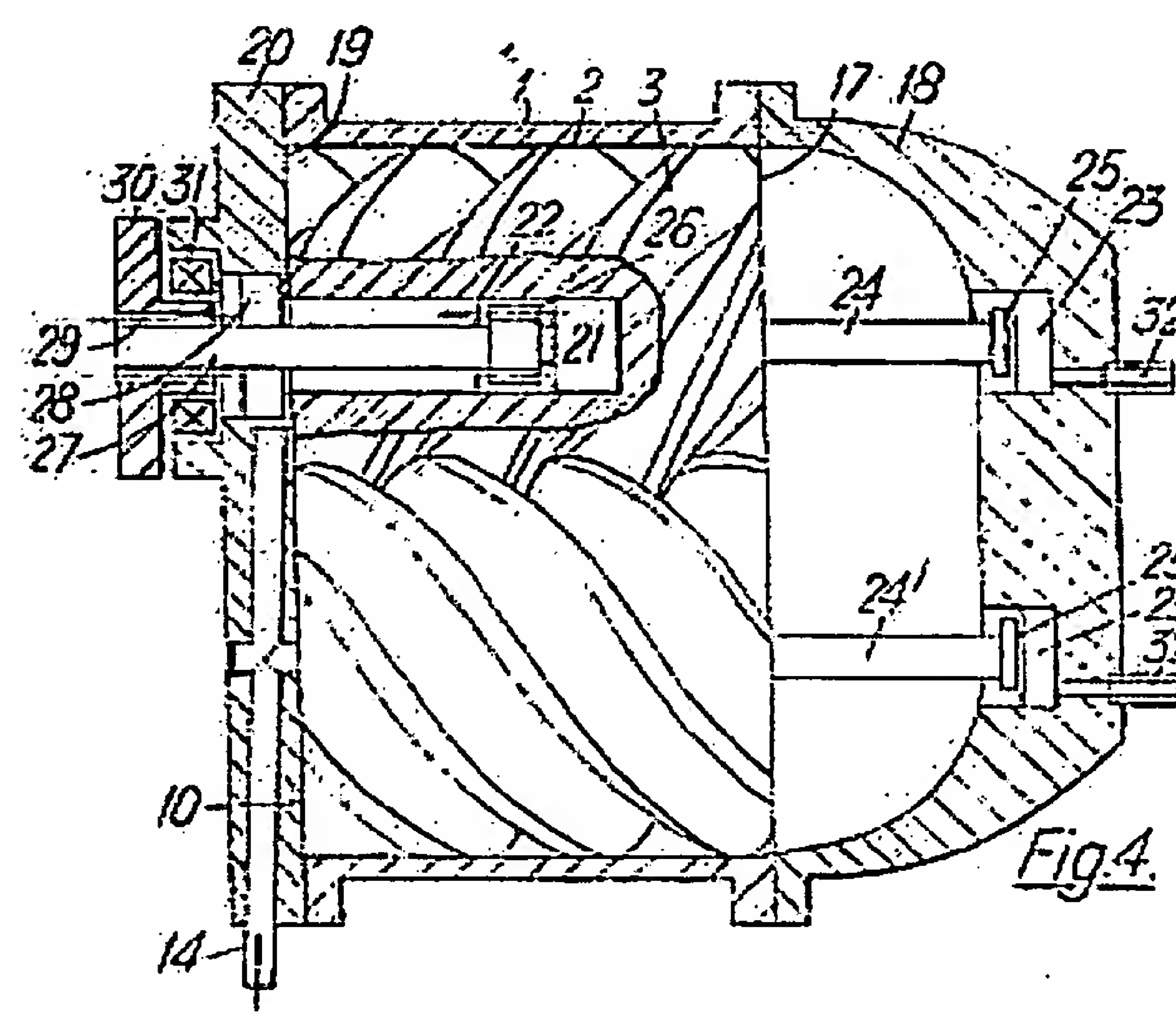

Fig.4.

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